# Modifications of a Rail/Wheel Wear Simulator to Measure Rail Curve Lubricant Performance

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## Abstract

*Wear of railroad rolling stock and rails costs millions of dollars each year in all rail systems throughout the world. The rail industry has attempted to address flange wear using lubricants. The choice of rail curve lubricant is currently not based on a standards approach. Measuring rail curve lubricant performance in a laboratory environment can address this information shortfall and reduce wear in rail rolling stock and infrastructure.*

*The equipment used to measure rail curve lubricant performance consists of two dissimilar disks, of matching contact profile, rotating in contact with one another. The outcome from the modifications was the ability to quantify rail curve lubricant performance using three performance criteria and to measure the changes in apparent viscosity with respect to accumulated strain damage.*

*This paper will provide a description of the equipment and modifications to the rail/wheel wear simulator and preliminary results to display the outcomes of the modifications.*

*Keywords: Rail Curve Lubricant Performance Measurement*

## 1. INTRODUCTION

In industry, attempts have been made to address wear using rail curve lubricants. There are presently a large number of lubricants and lubricant applicators used on existing rail networks. The choice of lubricant and applicator is currently not based on a standard of performance. The aim of this project was to convert a rail wear test device to measure lubricant performance in terms of decay rate, shear capacity and tractive power. This paper will describe the modifications and issues encountered in the conversion of a rail wear test device to measure rail curve lubricant performance.

## 2. Background

In the work of Clayton et al (1988; 1989) lubricants were investigated in both track and laboratory conditions. The testing was designed to measure the four features that the author proposed are important for flange lubrication: mobility; durability; lubricity; and contamination. The results of the testing yielded a very low correlation between field and laboratory. In addition to this low correlation it was found that the lubrication conditions in the two tests were different. Also the performance variation between the lubricants was questionable statistically. In summary Clayton et al (1989) states "At the present time, no laboratory test would appear to be able to be used with confidence to evaluate the in-service performance of wheel/rail lubricants."

Witte and Kumar (~1986) and Kumar (1991) designed a new test and apparatus for design of rail lubricants in response to an industry need for a standard test. Their focus, in terms of lubricant properties, was on lubricant mobility, durability and lubricity. Their work ignored the effects of lubricant migration that was investigated in the work of Clayton (Clayton, Danks et al. 1988; Clayton, Reiff et al. 1989). This new device focused on simulating the stress and creepage properties, which is in contrast to the work of Clayton (Clayton, Danks et al. 1988; Clayton, Reiff et al. 1989) which utilised an existing wear test device.

Clayton et al (1988; 1989) concluded that the new test correlated with a larger scale wheel/rail simulator, but quantitative correlation with field data was not performed. Qualitative comparison between the laboratory and field data yielded some correlation but the results were inconclusive. In summary the results of this work allowed for the preliminary analysis of lubricants with respect to the parameters relevant to the wheel/rail system.

## 3. Description of equipment (Rail/Wheel Wear Machine)

The rail/wheel simulator developed and used for this research originated from the BHP Melbourne Research Laboratories in Australia. This machine was purpose built by the laboratories to investigate wear of rail/wheel couples (Marich and Mutton 1989). BHP Billiton is a major supplier of materials to the rail industry and conducts their own heavy haul rail operations.

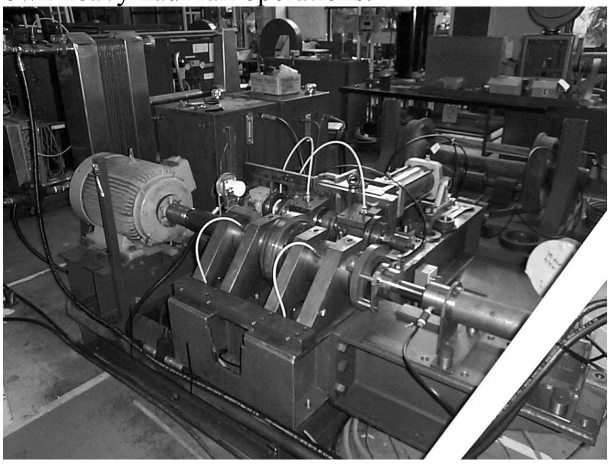


**Figure 1 – Rail/wheel simulator**

In its original form the rail/wheel simulator was used to test wear rates of rail/wheel couples. These couples consisted of different grades of rails and wheels, from those currently in use to laboratory prepared samples. The prepared samples had a range of hardness, chemical compositions and heat treatments. Results from the wear machine were used to compare materials varying in both strength and hardness. Wear rates were also measured for continuously lubricated conditions, which is where the importance of this equipment lies for the current thesis.

The equipment used to measure rail curve lubricant performance consists of two dissimilar disks, of matching contact profile, rotating in contact with one another. The first disk, a simulated rail, is driven by an electric motor which then drives the second wheel, a simulated railroad wheel, through the contact. The simulated railroad wheel is hydraulically braked to simulate the traction under rolling conditions.

The variables of the simulated contact that are controlled with this equipment are contact stresses, input and output disk speeds, slip ratio between disks, disk geometries and material properties, and lubricant types including biodegradable products.

The modifications to the rail/wheel simulator were carried out to convert the device from one designed to measure wear, to one that can measure rail curve lubricant degradation.

The wear test machine has two load parameters. Vertical load, the first parameter, simulates the axle load of the system. In the centre rear of the photograph in Figure 1, the vertical load pneumatic ram can be seen. The second parameter, lateral load, simulates curvature (flange contact). In the front right of the photograph in Figure 1, the lateral load pneumatic ram can be seen.

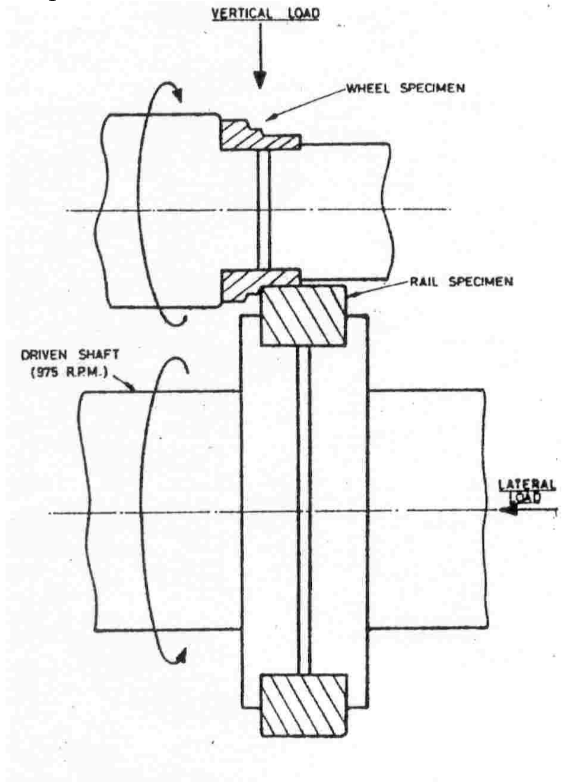



**Figure 2 Loading Diagram for wear investigation of Marich and Mutton(1989)**

These loads are depicted in Figure 2 from Marich and Mutton (1989). This arrangement is suitable for imitating a range of loading conditions, such as those experienced in the field.

One of the loading conditions, slip percentage, is important to determine the velocity profile across a given contact area. The slip percentage of the samples used in the work of Marich and Mutton (1989) was approximately 20%, compared to a maximum value of approximately 5% from a real rail/wheel system.

The original simulated system of Marich and Mutton (1989) and modified rail/wheel simulator used in this project is flexible to allow the use of a variety of wheel and rail profiles. These profiles can be taken from new design drawings or profiles of worn rolling stock and then be machined into the blank samples.

## 4. Equipment Modifications

The section on equipment modifications broadly outlines the deficiencies for lubrication research in

the simulator and the steps taken to address them.
The rail/wheel simulator was originally configured for rail/wheel wear investigation. Communication with the previous researchers (Marich and Mutton 1989) highlighted some design deficiencies of the equipment that were then rectified for the purposes of this research project. Significant modifications were then proposed and implemented in heat dissipation, tread load and the data acquisition systems to carry out the objective of quantifying rail curve lubricant performance through laboratory simulation. The modified configuration used in this paper can be seen in Figure 3.

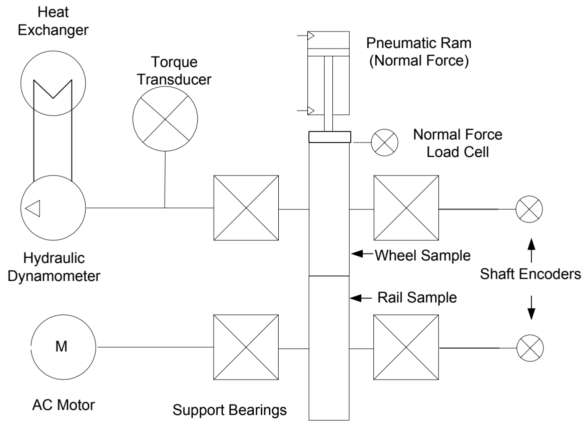



**Figure 3- Schematic diagram of the rail/wheel simulator.**

## 4.1. Heat Dissipation

The primary concern with the operation of the rail/wheel simulator in the previous research of Marich and Mutton (1989) was the dissipation of heat generated during testing. Heat is added to the system through sliding friction of the samples, friction in the rolling element bearings of the sample holders and from the hydraulic dynamometer. Heat is transferred to and from the simulator through the hydraulic system which is used for lubrication and the dynamometer (to apply tractive force). The hydraulic dynamometer is sensitive to variations in the oil temperature. It is therefore important to maintain a moderate and constant oil temperature. In this case the oil temperature was limited to between 40˚C to 45˚C to reduce the friction forces in bearings lubricated with the hydraulic oil.
The original heat exchanging system consisted of coiled copper tube wound in a continuous loop in the main oil storage tanks through which water flowed. It was assumed that the input power from the variable frequency drive is converted to heat, with minor losses to noise and vibration. Given this information it was believed that the heat exchanger was required to have a 22 kW minimum capacity. A new heat exchanger was therefore required as the existing heat exchanger possessed a capacity of approximately 2 kW.
The analysis of the heat transfer system, using the methods of Holman (1997), identified that a water-oil plate heat exchanger was suitable. This type was selected due its high surface area to volume ratio, compact design and low fluid resistance. These advantages were essential when considering the overall heat transfer coefficient of oil-water compared to water-water is at least an order of magnitude smaller.
The plate heat exchanger used was originally commissioned for a milk-water application and in an oil-water application capacity was predicted at 100 kW. With the plate heat exchanger installed and at full heat load, the outlet water of the heat exchanger is imperceptibly hotter than at the inlet when operating at full capacity.

## 4.2. Loading Mechanisms

The original loading mechanisms utilised a mechanical screw to apply the load, see Figure 4. Inside the screw a rubber cylinder was located between the end of the screw and the load application point (within the encasing indicated). The measured tread load in this configuration varied widely as a result of vibrations. In addition, the applied load would change with temperature and from the viscoelastic properties of the rubber spacer.

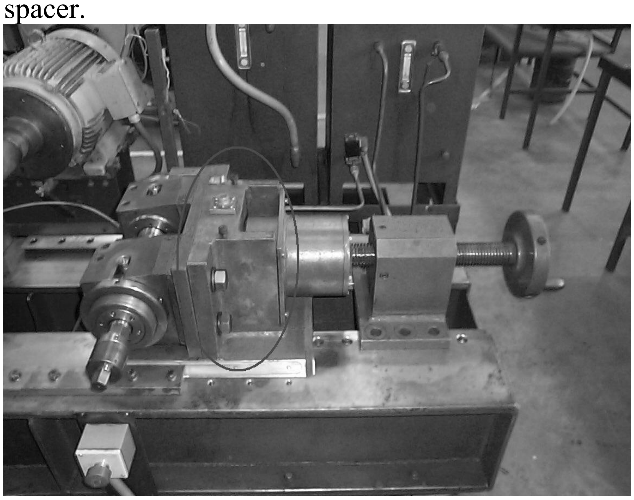


**Figure 4 – Tread loading mechanism showing original screw force applicator**

The screw load applicator relied on displacement to provide a force. For a constant force the screw load applicator required a fixed displacement and constant temperatures. These considerations led to a redesign of the tread loading mechanism.
A pneumatic system for tread loading was

designed to replace the existing mechanical system. The installed system consists of a high capacity pneumatic source supplying a single pressure control valve which supplies a pneumatic actuator. The advantages of the pneumatic system are that it reduces the amplitude of loading oscillations and reduces load variations.

Further testing of the modified loading system identified that normal force increased over the course of a test. The cause of which was found to be the pneumatic regulator which does not sense downstream pressure, therefore did not reduce the normal force to the set point. Increase in pressure cannot be dissipated through the pressure control valve but must be bled off through another mechanism, in this case a controlled system bleed. The bleed was tuned to provide the minimum amplitude of load oscillation.

The effects of loading modifications have been twofold. The variation of load with respect to time has been measured with the new configuration and found to be negligible whereas previously this was a serious concern. Previously constant changes to the mechanical system were required to maintain the test load. Outcomes of increasing load stability and system control have been achieved with this modification.

### 4.3. Wheel Sample Holder Alignment

The wheel sample head has three bearing surfaces upon which friction acts. This friction modifies the load experienced at the contact between rail and wheel samples and is accounted for in the calibration methodology. Two of the surfaces are equipped with Teflon-bearing pads which are then lubricated with oil to provide low friction contact surfaces. The third surface was originally a steel-steel contact and is now a lubricated brass-steel contact.

In its original configuration the wheel sample head could rotate about its vertical axis in the running channel and apply pressure at the corners of the guide block into the channel walls. The pressure in the corners would increase from thermal expansion of the wheel head as the test progressed. The effect of this was to reduce the sensed/measured load while applying an increasing tread load to the rail and wheel samples. The problem was detected by using a measured extraction force. Mechanical advantage in the mechanical screw loading system may have masked the appearance of this phenomenon.

Identification of the steel-on-steel bearing surface also highlighted the need for improved load alignment. An investigation showed that alignment issues existed as plastic deformation and associated wear accumulated on one end of the test samples during a commissioning test. The increased plastic deformation was discovered by increased hardness at the edge of wheel and rail samples. Rectification of the alignment was carried out through guide rail adjustments.

The force to move the wheel sample holder can be a component of the measured applied force from the pneumatic ram under static conditions, a source of error in this measurement. The friction force remains after moving the wheel sample holder into position. During testing the vibrations through the contact patch into the wheel sample holder caused a reduction in the friction force, such that it approached zero and the measured normal force is the applied normal force. The friction force arose from the two normal forces, the force of gravity from the mass of the sample holder and the force from the applied braking torque. A variation of the force will cause a corresponding variation in the friction force to move the wheel sample holder. In addition, as the wheel sample holder expands within the alignment rail channel, from an increase in temperature, the friction force may increase. A maximum friction force of less than 500N was required. It was therefore necessary to know the tolerance of the alignment rails as this is the only adjustment available to reduce the friction force in moving the wheel sample holder.

Experimental measurement of the tolerances of the alignment rails was carried out by measuring the force required to move the sample head in both directions and the tolerance gap set to an acceptable friction force of 300N. Verification, under a range of operating temperatures, that the friction force was less than 500N under test conditions was carried out and found to be less than the threshold value of 500N (approximately 350N).

### 4.4. Data Acquisition

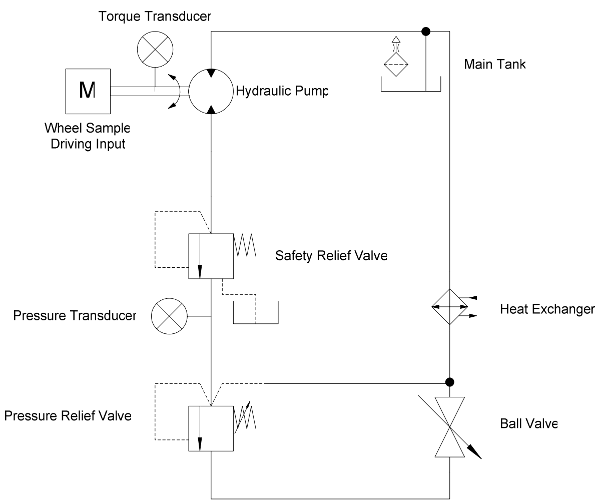
In the process of checking the tread loading system, deficiencies in the load measurement system became apparent. High frequency and resolution data were acquired from the original load cell electronics to find the source of the issues. Harmonic noise was found. The load cells were sensitive to electronic noise from the 50 Hz single phase supply power from the variable frequency drive.

The load cells were a custom design of a membrane type installed between the loading mechanism and the wheel sample holder. Two changes were made, addition of a 50 Hz filter to the electronics and installation of an earthing circuit. Verification by repeat data collection showed elimination of the power supply noise at 50 Hz and reduction to barely detectable levels of the variable frequency drive noise. Mechanical vibration of the entire simulator was now the dominant component of signal noise in the load cell measurement.

### 4.5. Tractive Force Application System

A hydraulic dynamometer system was installed to apply and control the tractive/shearing force between the rail and wheel samples. The dynamometer system controlled braking torque only, contrary to other twin-disk systems which control slip and measure friction force (Marich and Mutton 1989; Markov 1995; Tyfour, Beynon et al. 1995; Olofsson and Telliskivi 2003). The limitation of controlling braking torque is high slip conditions, which occur at the beginning of lubricated tests.

**Figure 5 – Hydraulic dynamometer system.**




The hydraulic system re-design was in part due to the change in test sample shape from the work of Marich and Mutton (1989). Previously the wheel samples had a flange upon which a load was applied.

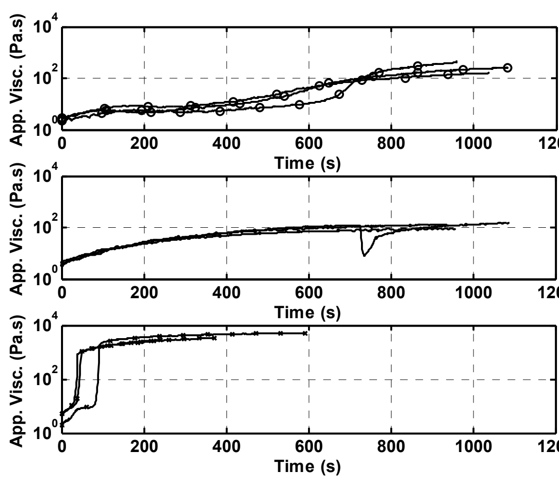
### 4.6. Slip/Creep Measurement

The focus of the research was to characterize rail curve lubricants, with the approach of measuring accurately the slip or shearing conditions at the contact between the lubricated rail and wheel samples. The rotational speed measurement system has been totally redesigned to increase the resolution of speed and slip measurements/calculations of the rail and wheel samples. Previously, in the work of Marich and Mutton (1989), the speed of shaft rotation, for rolling speed calculation, was measured with a inductance proximity probe and notched wheel. This method lacks the resolution and accuracy required to measure slip and small changes in slip (<1%). To address the issues with the original inductance probes, shaft encoders were installed. The encoders have a resolution of 5400 encoder counts per revolution to gather high resolution data for the shaft position and speed. This modification increased the accuracy of slip/creep measurement by two orders of magnitude.

## 5. Measurement Details

Methods for the measurement of the variables of interest are now discussed. Rotational speed was measured with shaft encoders for the purpose of measuring rolling velocity, sliding velocity and slip ratio. Output torque was measured with a torque transducer in the hydraulic dynamometer system for calculating output power, shear force, shear stress and power absorbed by a lubricant film. The variable frequency drive on the input shaft was used to measure input torque, for calculating input power and power absorbed by a lubricant film. Temperatures of the rail and wheel samples were measured during testing using a hand held infra-red thermometer. Normal load, important for calculating the stress distribution of the contact between rail and wheel samples, was measured using a calibrated force transducer.

**Figure 6 – Apparent viscosity versus time for Lubricant A(top), Lubricant B (middle), and Lubricant C (bottom) for a simulated 27.5 tonne axle load travelling at 42km/hr into a 300m radius corner.**




The combination of measurements allows for the measurement/calculation of performance criteria of the tested rail curve lubricants. These performance measurements are: apparent viscosity, a measure of the lubricity presented with respect to accumulated strain, see Figure 6; total absorbed energy, the energy absorbed in the lubricant film instead of being utilised for wear processes, see Figure 7; and total distance slid, the sliding distance or accumulated strain achieved prior to development of a set tractive force limit, see Figure 8. In all tests Lubricant A is the best performer will the largest absorbed energy and sliding distance and the longest period of reduced viscosity.

**Figure 7 – Total absorbed energy (27.5 tonne axle load travelling at 42km/hr into a 300m radius corner )**

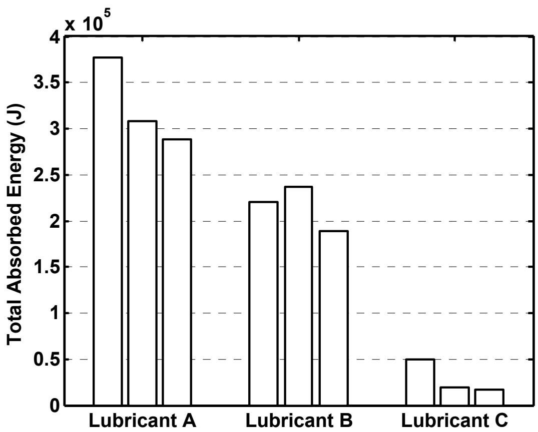
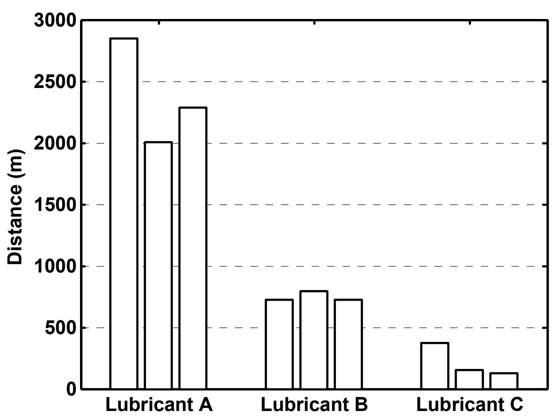



**Figure 8 - Total slid distance (27.5 tonne axle load travelling at 42km/hr into a 300m radius corner )**




## 6. Summary

Details of the modifications to the experimental device have been presented, highlighting the need for modifications and improvements to achieve the objective to quantify rail curve lubricant performance through laboratory simulation. Temperature of the rail/wheel simulator can be regulated with the installation of the plate heat exchanger to improve simulation conditions. Loading of the test samples has been modified with pneumatic rams to improve simulation by applying a more constant force. Data acquisition has been modified and improved across all measurements especially in the area of slip measurement. High resolution slip measurements were attainable using the newly installed shaft encoders for measuring lubricant film decay. Tractive force control was achieved through the installation of a hydraulic dynamometer.

Methods for rail curve lubricant performance measurement have been presented. These performance measurements are total absorbed energy, the energy absorbed in the lubricant film instead of being utilised for wear processes; total distance slid, the sliding distance or accumulated strain achieved prior to development of a set tractive force limit; half life of lubricant, the time taken for a lubricant to lose half of its sliding performance; and apparent viscosity, a measure of the lubricity presented with respect to accumulated strain. Using the new method of lubricant performance measurement and the modified wear simulator rail curve lubricant performance has been measured.

## 7. REFERENCES

Clayton, P., D. Danks, et al. (1988). "Laboratory Assessment of Lubricants for Wheel/Rail Applications." Lubrication Engineering **45**(8): 501-506.

Clayton, P., R. Reiff, et al. (1989). Wheel/Rail Lubricant Performance: In-Track and Laboratory Test Comparisons. The Fourth International Heavy Haul Conference, Brisbane, Queensland, Australia.

Holman, J. P. (1997). Heat transfer. New York, McGraw-Hill Companies.

Kumar, S., P. J. McDuff, et al. (1991). Development of a Laboratory Test for Rail Lubricants. NLGI 58th Annual Meeting, Phoenix, Arizona, USA.

Marich, S. and P. J. Mutton (1989). An Investigation of Severe Wear Under Rolling/Sliding Contact Conditions Existing in Heavy Haul Railways. 7th International Wheelset Conference, Vienna.

Markov, D. (1995). "Laboratory tests for wear of rail and wheel steels." Wear **181-183**(Part 2): 678-686.

Olofsson, U. and T. Telliskivi (2003). "Wear, plastic deformation and friction of two rail steels--a full-scale test and a laboratory study." Wear **254**(1-2): 80-93.

Tyfour, W. R., J. H. Beynon, et al. (1995). "The steady state wear behaviour of pearlitic rail steel under dry rolling-sliding contact conditions." Wear **180**(1-2): 79-89.

Witte, A. C. and S. Kumar (~1986). Design and Laboratory Testing of Rail Lubricants.